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VIBRATION INVESTIGATION OF HELICOPTER ENGINE COOLING FAN.(U)

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N. S. SWANSSON and G. A. DUKE

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by

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SUMMARY

An investigation into the cause of the unacceptably high incidence of fatigue failure of blades in the engine cooling fan fitted to Sioux helicopters led to a search for resonant frequencies of the fan blades. Tests on a stationary fan and in a rotating rig showed resonant vibration at various fan speeds; in particular, high stresses were generated by a second shaft order resonance close to the normal operating speed.

A modification to overcome the problem was proposed, consisting of a stainless steel shroud fitted over the blade tips. Tests showed that this modification reduced stresses to a negligible level, and the mechanical soundness of the scheme was confirmed by an endurance run followed by an overspeed test. The modified fan was fitted to a helicopter and measurements comparing its cooling performance with a standard fan showed only small differences.

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ABSTRACT

Tests conducted during an investigation into fatigue failure of blades of the engine cooling fan fitted to the Sioux helicopter showed that high stresses were generated by a second order resonance close to normal operating speed. A modified fan fitted with a tip shroud reduced these stresses to very low level. Performance and endurance tests on the modified fan compared satisfactorily with the original.

CONTENTS

1. INTRODUCTION

In the past few years an unduly high number of fatigue failures have occurred in the engine cooling fans on the Sioux helicopter (Bell Model 47G 3B1) and its equivalent versions, both military and civil. This fan provides cooling air to the engine, which is a Lycoming TVO 435 six cylinder horizontally opposed air cooled engine, having a rating of 230 HP at 3200 rpm. The fan is driven at 1·5 times engine speed through the main helicopter transmission. As a part of the engine installation rather than the basic engine assembly, responsibility for the fan appears to lie with the helicopter manufacturer.

Failure of a fan blade or blades would not of itself be immediately disastrous. However consequential damage is almost certain to result in total loss of engine cooling (frequently by destroying the V belt drive to the fan) followed by rapid overheating and engine seizure. Also the fan assembly is mounted in a narrow space between the engine and the helicopter cabin, with various control rods, pipes and other accessories located close to the fan. In one incident, the impact of a ruptured blade broke a bell crank in the control linkage, and pitch control of the rotor blades was lost. Fortunately this event occurred during ground running; had it occurred in flight the consequences would have been disastrous.

Since hazards arise essentially from secondary damage, helicopter users in the British Army have considered schemes for containing the blades after rupture. Australian users feel that the difficulties of blade containment, which is only a partial solution, do not warrant the effort involved, and are seeking a basic solution to the problem whose major aspect is one of flight safety.

2. BACKGROUND

The fan disc (Fig. 1) has 16 blades pressed from aluminium sheet. Originally three stiffening discs were bolted to the front face of the fan, with an outer circle of 12 bolts having load sharing washers bearing on the rear face of the fan disc, and an inner circle of 6 bolts which attached the assembly through a spacer to a cast aluminium hub. The fan is driven by twin V belts, the belt pulley being bolted separately to the hub, and has a normal operating speed of 4800 rpm.

In the period between mid 1969 and mid 1971 twelve fatigue failures of fan blades were reported, at times since overhaul between 99 and 590 hours. The fan is not "lifed" but assessed on condition at overhaul, and total lives generally are not known. In all but one of these cases, crack initiation was attributed to stress corrosion, the indentations of the washers in the rear of the fan disc proving a particular location of weakness. An investigation was undertaken by Hawker de Havilland in 1971 (Ref. 1), comprising essentially measurements on a stationary fan of principal natural frequency and damping, using impulse excitation. This led to a modification whereby two stiffening discs were fitted to each side of the fan, the inner discs being bonded to the fan and the outer ones bolted (to promote damping). The fan with four discs was then fitted to the hub without the original spacer.

The extra disc and bonding stiffened the fan and increased its natural frequency about 12% compared with the original design, and roughly doubled the damping coefficient. The modification has since been implemented at fan overhaul, but operationally it has not been successful, since, so far as evidence is available, the failure rate of modified fans is at least as great as before modification. Crack initiation occurs mainly at the pre-existing washer indentations, but has occasionally been observed at the trailing edge blade fillets, with a time since overhaul in one case as little as 18 hours.

As noted above, helicopter users outside Australia have also experienced fatigue failures of fan blades, and Westland Helicopters (Ref. 2) conducted an investigation in late 1973. This consisted of a search for resonant frequencies and determination of corresponding mode shapes, with particular reference to coupling between disc and blades, together with a survey of vibration induced stresses. All measurements were made on a stationary fan; the effects of fan rotation were indicated qualitatively but not predicted quantitatively. The report concludes that rotating

tests either on a rig or in the helicopter installation would be needed to positively identify the vibration mode, locate the exciting source and propose a remedy.

3. TESTS ON STATIONARY FAN

Despite the limitations of stationary measurements just noted, such tests are easily conducted and may be used together with theoretical formulae to predict behaviour of a rotating component. A resonance search was conducted using two different types of excitation; firstly by shaking the fan in the axial direction and secondly by air pulse excitation applied to a single blade. Blade motion was measured by a non contacting capacity probe, and stroboscopic illumination was also used to assist in determining mode shapes. Results using the vibration exciter are given in Table 1.

TABLE 1. Resonant Frequencies and Modes

(Fan S/No. A34-07541 Unmodified)

Frequency Hz	Mode
97–110	1st Flap
458	1st edgewise
578	1st torsion
595	
622	2nd flap?
1120–1130	3rd flap?
2105	not identified
3700	not identified

With the air pulse exciter, 1st flap modes at 103 and 109 Hz were observed, but other modes could not be excited (maximum pulse frequency during these tests was limited to 200 Hz). Another fan, Serial No. A34-6490 was later tested on the vibration exciter and its first flap frequencies of 104–116 Hz were about 7% greater than those in Table 1.

The different resonances observed for the first flap mode are the result of coupling between disc and blades, a topic examined in detail by Ewins (Ref. 3). Individual measurements of blade amplitudes of the 103 Hz and 109 Hz modes indicated that the former mode had 3 nodal diameters, while the latter had alternate blades in opposite phase and could be regarded as having 8 nodal diameters. In this case however only about half of the blades were vibrating with substantial amplitude with the remainder nearly stationary. The modes observed are similar to some of the various modes predicted by Ewins for a disc with detuned blades; in practice only some of the theoretically predicted modes can be excited.

The increase in natural frequency in a rotating fan caused by centrifugal stiffening of the blades was estimated using the results of Dokainish and Rawtani (Ref. 4). Taking the fundamental flap range of frequencies as 97 to 110 Hz, the range of blade natural frequencies at a rotational speed of 4800 rpm was estimated at 167 to 175 Hz. The formula used represents the blade as a uniform rectangular cantilever plate encastré at the radius of attachment to the disc. Compared with this model, actual blade stiffness will be increased because of its camber, and reduced because of the disc flexibility and the narrower width of the untwisted part of the blade next to the disc. These effects tend to cancel one another and should result in a fair estimate of natural frequency being obtained.

4. SHROUDED FAN

The predicted rotating fan blade frequency of 167 to 175 Hz is very close to twice the operating speed of 4800 rpm (80 Hz), and it was believed that fan blade fatigue was caused by resonant vibration at or near this speed (a conclusion advanced tentatively in Ref. 2). It was decided at this stage to initiate a modification to the fan pending the commissioning of a rotating rig to verify the theoretical predictions of rotating fan blade frequency. The modification con-

sisted (Fig. 2) of a stainless steel shroud ring fitted over the blade tips, which were first machined so that the ring would maintain the original overall diameter. The shroud was retained by thin stainless steel plates rivetted to the blades close to their tips, and welded to the shroud itself.

A shroud substantially limits the vibration modes which will be produced by excitation of a given shaft order. Because the blade tips are constrained to move together, modes with adjacent blades in opposite phase will be suppressed. At the same time the 'umbrella' mode with all blades in phase, and the 'tilting' mode with one nodal diameter will be reduced in frequency since, although the stiffening effect of the shroud will slightly increase the blade natural frequency, the extra mass loading will tend to reduce it and the latter effect will dominate in these modes. Ewins (Ref. 3) gives a detailed description of coupled modes in a detuned blade-disc system, and the effects of a shroud in suppressing the excitation of particular modes.

Resonance search and mode shape measurements on the shrouded fan were made using the methods already described, and results are given in Table 2. The apparent anomaly that the one nodal diameter or 'tilting' mode has a lower frequency than the 'umbrella' mode with no nodes is associated with the proportions of elastic strain energy due to bending and to stretching of the disc; this was first discussed by Southwell (Ref. 6).

TABLE 2. Resonant Frequencies and Modes

(Fan S/No. A34-07541 Shrouded)

Frequency Hz	Mode
62	1st Flap 1 Nodal Diameter (Tilting)
94	1st Flap In Phase (Umbrella)
112	1st Flap 2 Nodal Diameters
225	1st Flap 3 Nodal Diameters
422	1st Flap 4 Nodal Diameters
638	1st Flap 6 Nodal Diameters
723	1st Flap 8 Nodal Diameters
1240	Not Identified

5. ROTATING TEST RIG

A small fan dynamometer rig, used previously for testing car and truck fans, was recommissioned for these tests. It comprised (Figs 3 & 4) a 20 HP variable speed DC motor driving the fan shaft by four V belts at a maximum speed just over 4800 rpm. Strain gauges were attached to the front and rear of the fan blade at its root, and connected to measure blade flexure stresses. Strain signals were taken out through water lubricated slip rings at the front of the fan. The rig simulated as far as possible the normal fan installation; the fan was surrounded by a fixed shroud supported by 4 struts on the downstream side, a large board simulated the helicopter cabin and the slip rings and their support bracket were designed to generate wakes giving an aerodynamic interference upstream of the fan similar to that produced by the V belts and other equipment in the helicopter installation. A mesh restriction was placed over the fan outlet so that it operated against a back pressure of the correct order.

A block diagram of the instrumentation used is shown in Fig. 5. Signals from the strain gauge bridge were fed to a differential amplifier (to eliminate common mode pickup) and were monitored by oscilloscope. The oscillating part of the strain signal was measured by an RMS voltmeter, having an analogue output which drove the Y axis of a plotter. A speed signal for the X axis of the plotter was provided by a DC tachogenerator on the fan shaft, and speed was also monitored by a photoelectric pickup and electronic counter.

6. ROTATING TESTS—UNSHROUDED FAN

A typical plot of vibratory strain against fan speed for the fan S/No. A34-6490 which did not have a shroud fitted is shown in Fig. 6a, confirming the expectation that there would be a

resonance between blade first flap frequency and second order fan shaft excitation, close to the fan operating speed. The table below gives resonances up to the 12th order which could be observed with the oscilloscope; resonances up to 8th order can be resolved on plots such as Fig. 6a.

TABLE 3. Rotating Fan Test

Resonances of Unshrouded Fan Ser. No. A34-6490

Fan Speed rpm.	Vibration Frequency Hz	Excitation Order
570	114	12
900	120	8
1218	122	6
1452	121	5
1836	123	4
2568	128	3
4608	154	2

By extrapolating the results of Table 3, the natural frequency of the fan at 4800 rpm is estimated at 157 Hz, somewhat below the range of 167 to 175 Hz predicted from static test measurements.

Fig. 6 shows that the fan tested at its normal running speed of 4800 rpm, is operating on the upper flank of the second order resonance and would possibly perform satisfactorily. But excessive vibration and fatigue failure is likely in a proportion of fans either because their natural frequency may be slightly higher than that of the fan tested (static tests on two fans gave a 7% difference in natural frequency) or because the engine (and fan) is operated for at least part of the time below its specified speed (precise trimming of which is the pilot's responsibility). Because of these factors, fatigue performance will be highly variable.

A number of variations in the test configuration were tried, to see what effect they had on the stress amplitude at resonance. The fan was run with and without the restriction mesh at outlet, a cross wind of about 20 m/s was directed across the face of the fan, and the angular position of the 4 downstream shroud struts was varied relative to the (simulated) upstream V belt wake interference. On the whole these variations did not significantly alter the resonant stress amplitude values—see Figs 6(b), 7(a) & (b).

It should be noted that the stress amplitude of about 48 MPa (7000 psi) cited in Figs 6 & 7 is measured at the centre of the blade at the root. This location was chosen for the strain gauges because strain gradients are low and a good comparison should be possible between shrouded and unshrouded fans. However substantially higher stresses could be expected around the fillet at the blade root (and other stress concentrators such as washer indentations) which could cause fatigue failure.

7. ROTATING TESTS—SHROUDED FAN

Tests similar to those on the unshrouded fan were conducted on the fan fitted with the shroud, and results are presented in Figs 6(c), 8 & 9. Only one resonance is present, the fan speed being 4300 rpm and resonant frequency 858 Hz, which is 12th shaft order. (With a 16 blade fan, it is curious that this frequency does not appear also as 16th shaft order at a lower fan speed). Once again the effect on resonant stress amplitude of cross wind, fan outlet restriction, and variation of shroud strut angle was investigated, and found to be relatively small.

The observed resonance is sharp, its stress amplitude is about 1/5 that of the unshrouded fan, and it is reasonably far removed from the normal fan operating speed of 4800 rpm. Efforts were made to identify the mode involved, to guarantee that it would not adversely affect the design. Preliminary calculation indicated that it could be due to vibration of the shroud, in a mode with 8 circumferential waves and nodes at each blade tip. To check this, a strain gauge was fitted to the shroud, and the two strain gauges on the blade were connected in opposite

instead of adjacent arms of the bridge, to measure blade tension rather than flexure. In both cases the measured amplitudes of stress oscillation were negligible indicating that no significant vibration was occurring in the shroud. A static test with the air pulse directed at the shroud midway between blades also failed to excite any significant resonance in the shroud.

The results of Table 2 indicate that the resonance is probably a first order flap mode with eight nodal diameters having a frequency of 723 Hz with the fan stationary; centrifugal stiffening would increase this to the observed frequency of 858 Hz when rotating.

8. FURTHER TESTS OF INTEGRITY

To substantiate the durability of the shrouded fan an endurance test was conducted, in which the fan was run at resonance speed of 4300 rpm for 20 hours, equivalent to about 6×10^7 vibration cycles. After the run the fan was crack tested with fluorescent dye penetrant around the blade roots and shroud welds; no indication of cracking was found. Finally the fan was mounted in a spin rig and oversped 22% to 5850 rpm without any detectable deterioration.

9. FLIGHT TESTS

Rig tests indicated that fitting a shroud would overcome the basic vibration problem without compromising the mechanical soundness of the fan. It was considered desirable also to instal the modified fan in a helicopter to compare its cooling performance with a standard fan.

Circular pressure manifolds upstream and downstream were used to measure the pressure rise across each fan under similar flight conditions. It was found that the static pressure rise across the modified fan was about 20% lower than the standard fan.

Engine cylinder temperatures were measured at the same time using the aircraft instruments. At ground hover and maximum cruise conditions temperature increases were higher by 6·8% and 4% respectively with the modified fan, while at normal cruise condition temperatures were essentially the same with both fans. Averaged over all tests the temperature increase was 3·5%.

An empirical relation (Ref. 5) between the cylinder temperature rise ΔT and the pressure of cooling air above ambient ΔP, has the form:

$$\Delta T \propto \Delta P^{-0 \cdot 2}$$

The observed average changes in temperature and pressure agree approximately with this relation.

Considering all results, it appears that the fan modification is likely to cause an increase in engine temperature of about 5°C averaged over different flight conditions.

10. CONCLUSION

It has been established that a major resonance occurs at the fundamental flap frequency of the fan blades with an excitation of twice fan shaft order, at or close to normal fan operating speed. This resonance is responsible for fan blade fatigue failures. Since the severity of resonance depends on small unpredictable variations in fan blade frequencies, and upon how precisely engine operating speed is controlled, occurrence of fatigue will be erratic.

One fan has been modified by fitting a shroud around its periphery. The modified design eliminates all except one minor resonance, and this is reasonably removed from the operating speed. An endurance test at the resonance condition was conducted, confirming the integrity of the modification.

The performance of the modified fan has been compared with that of a standard fan under flight conditions with results showing a small decrease in pressure rise and consequently a small rise in engine temperature.

REFERENCES

1. B. Tydeman	Investigation into Failures of Cooling Fan Assembly P/N 47-661-801-19 as Fitted to Sioux 47G-3B1 Helicopters, Hawker de Havilland Australia Pty. Ltd., Engineering Report No. Tech. 69, 1971
2. R. J. W. Waplington	Sioux Helicopter—Engine Cooling Fan—Non Rotational Natural Frequency Investigation, Westland Helicopters Ltd., Test Report G.3.4072.1, 1973
3. D. J. Ewins	Vibration Characteristics of Bladed Disc Assemblies, J. Mech. Eng. Sci. Vol. 15 No. 3, 1973, p. 165
4. M. A. Dokainish and S. Rawtani	Vibration Analysis of Rotating Cantilever Plates, Int. J. Num. Meth. Eng. Vol. 3, 1971, p. 233
5. A. W. Judge	Aircraft Engines Vol. 1, Chapman & Hall, 1940
6. R. V. Southwell	On the Transverse Vibrations of a Uniform Circular Disc Clamped at its Centre; and on the Effects of Rotation, Proc. Royal Soc. Vol. 101, Series A 1922, p. 133

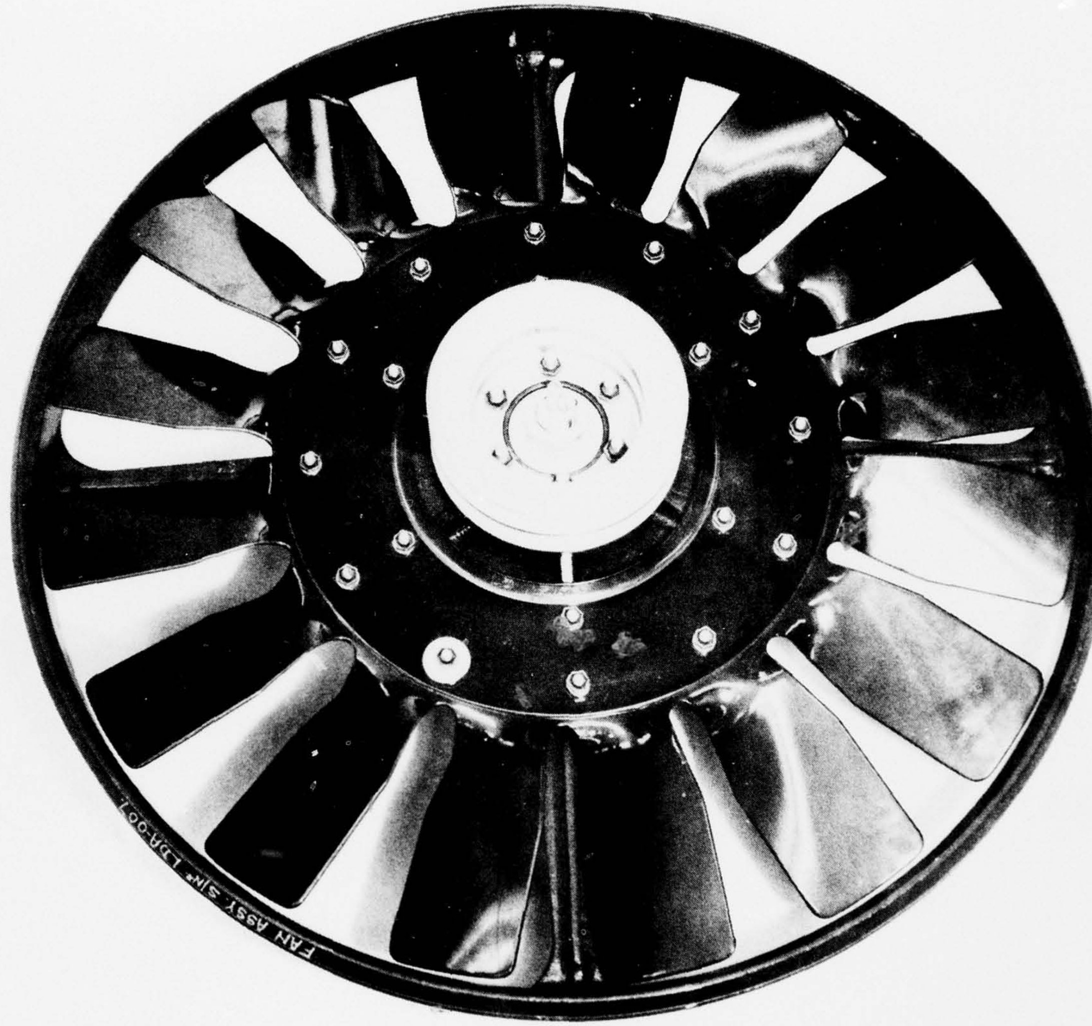

FIG. 1. THE COOLING FAN ASSEMBLY – FRONT FACE

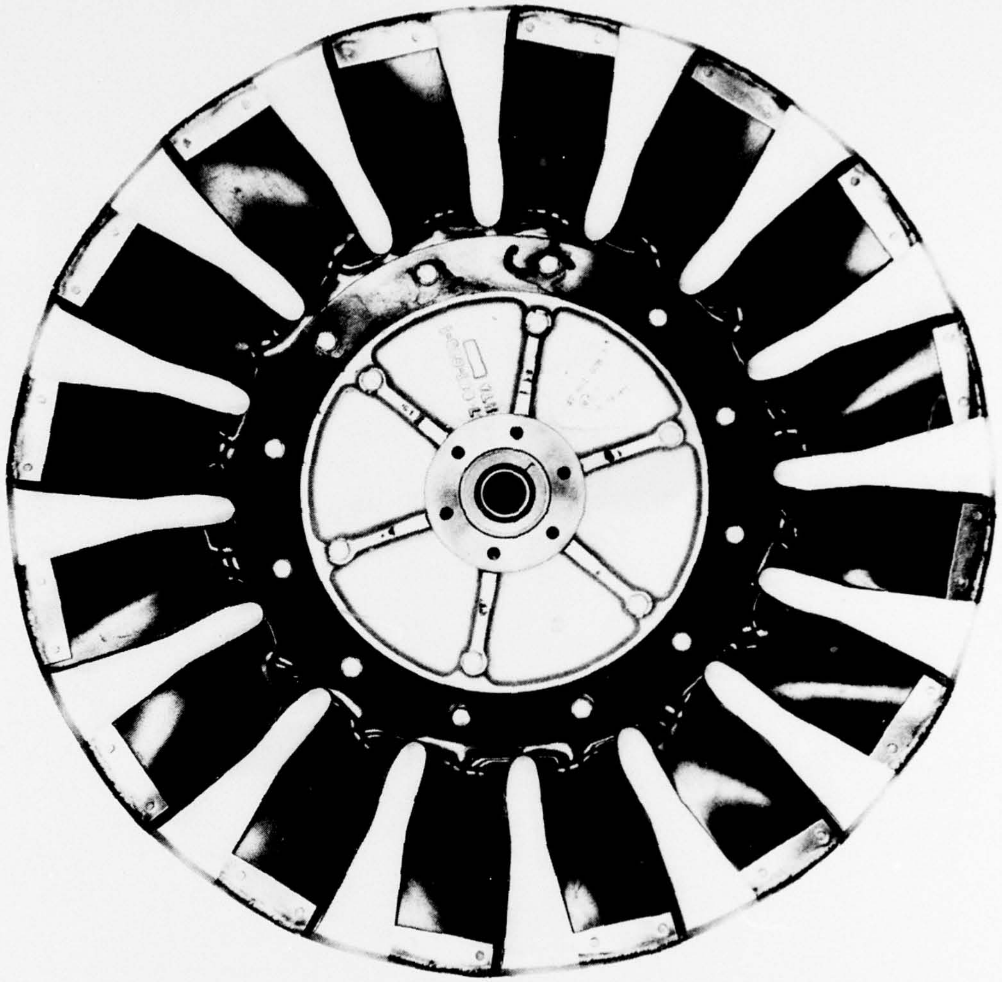

FIG. 2. THE SHROUDED FAN FROM DOWNSTREAM SIDE

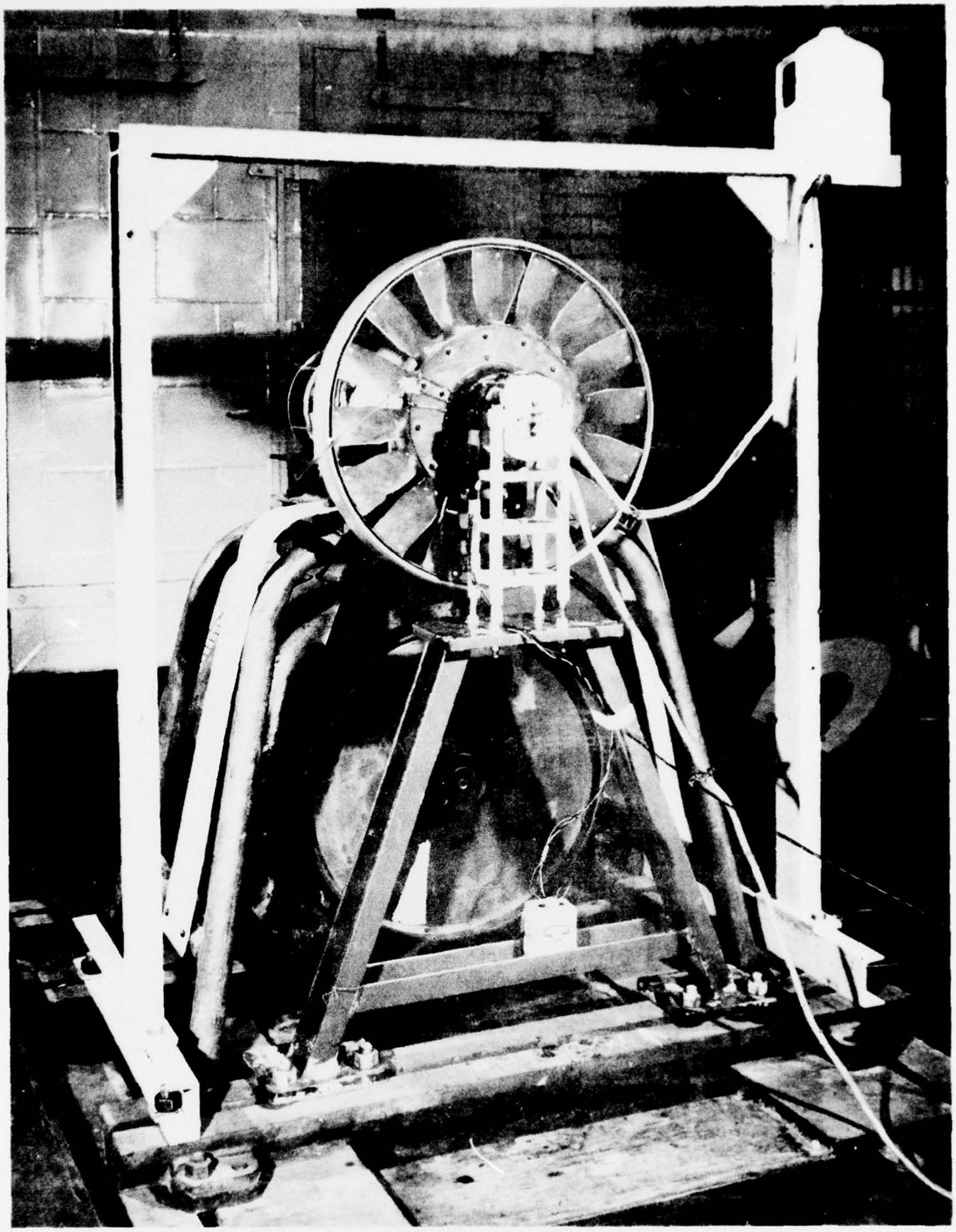

FIG. 3. THE FAN TEST RIG UPSTREAM VIEW

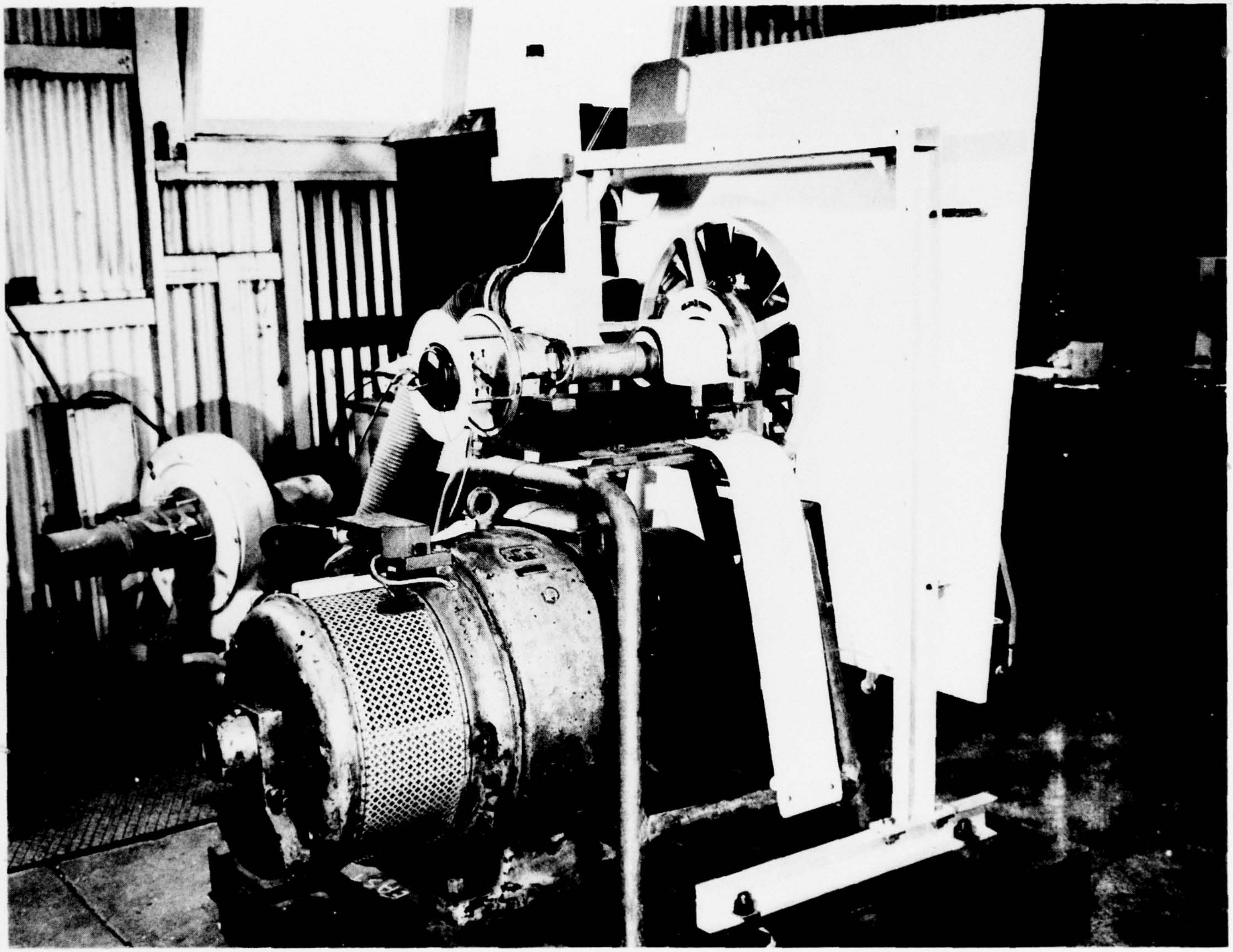

FIG. 4. THE FAN TEST RIG DOWNSTREAM VIEW

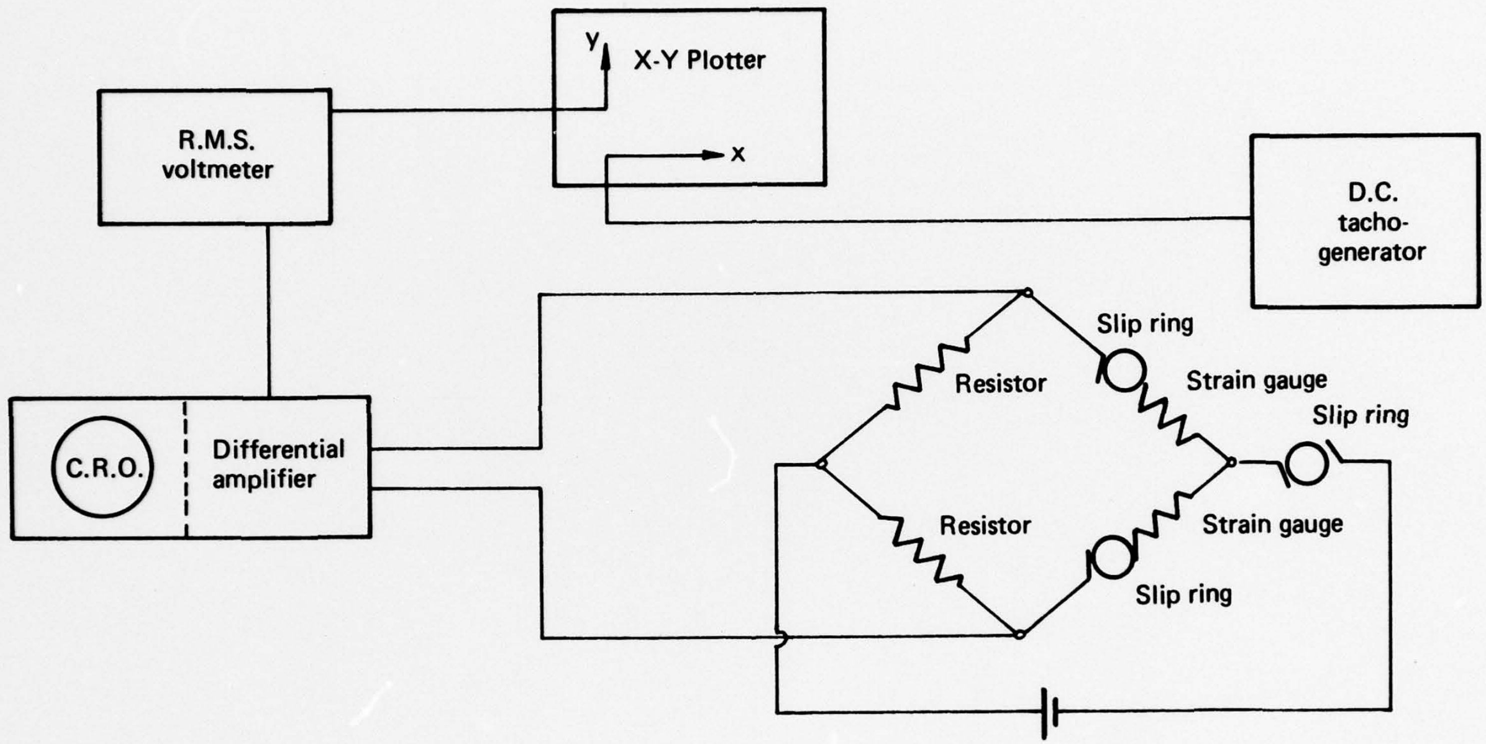

FIG. 5. DIAGRAM OF INSTRUMENTATION

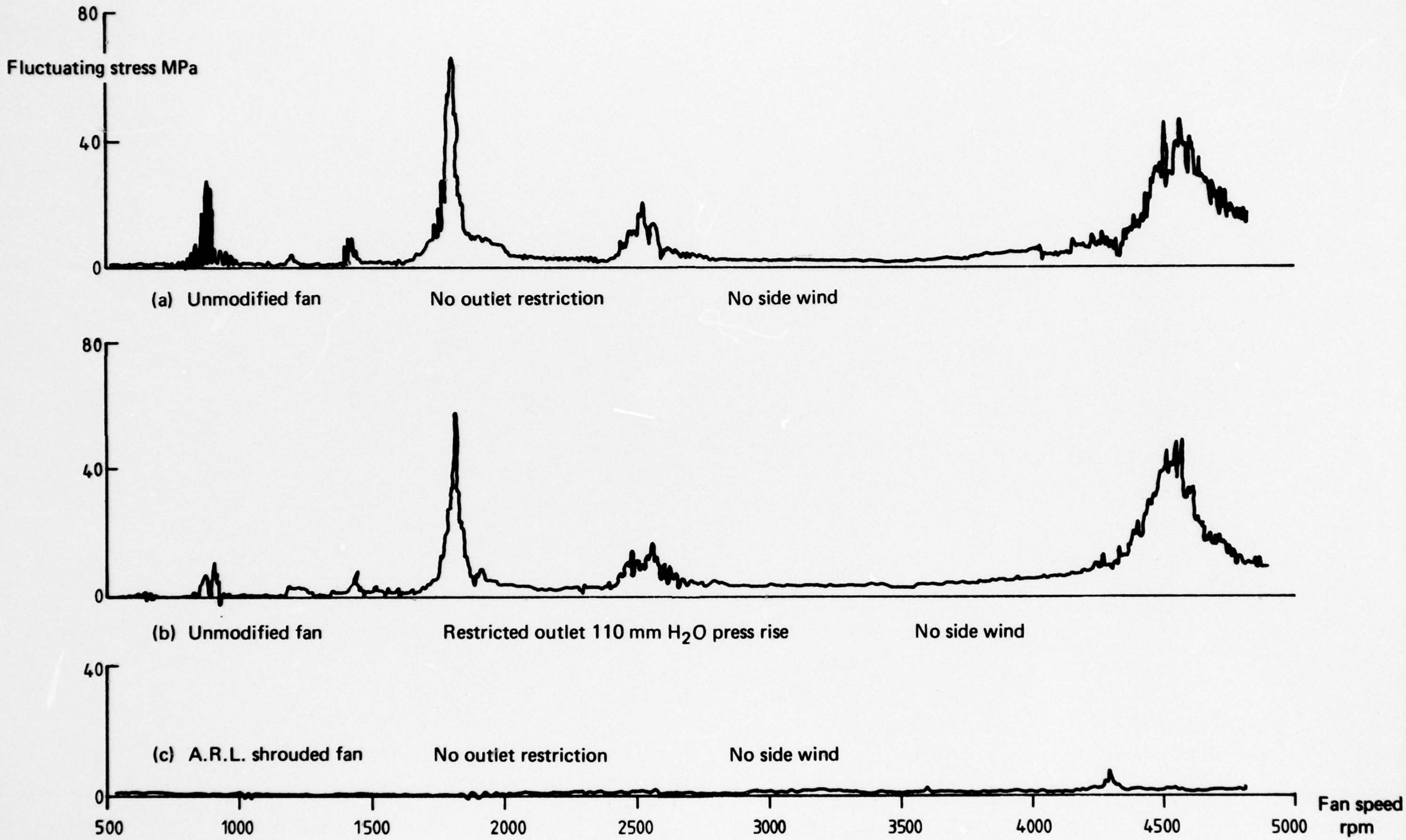

FIG. 6. FLUCTUATING STRESS IN BLADE ROOT, UNMODIFIED AND SHROUDED FANS

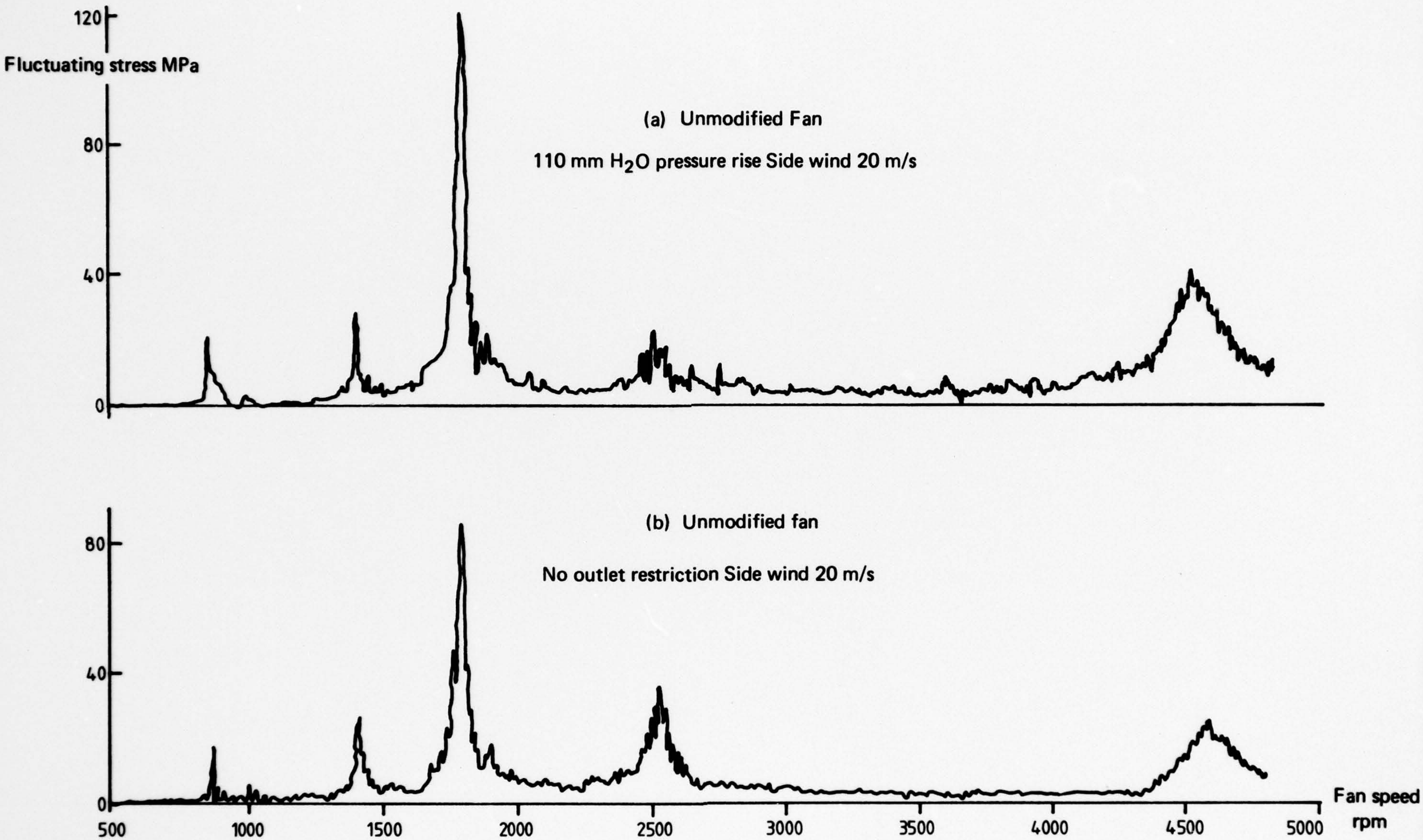

FIG. 7. FLUCTUATING STRESS IN BLADE ROOT OF UNMODIFIED FAN

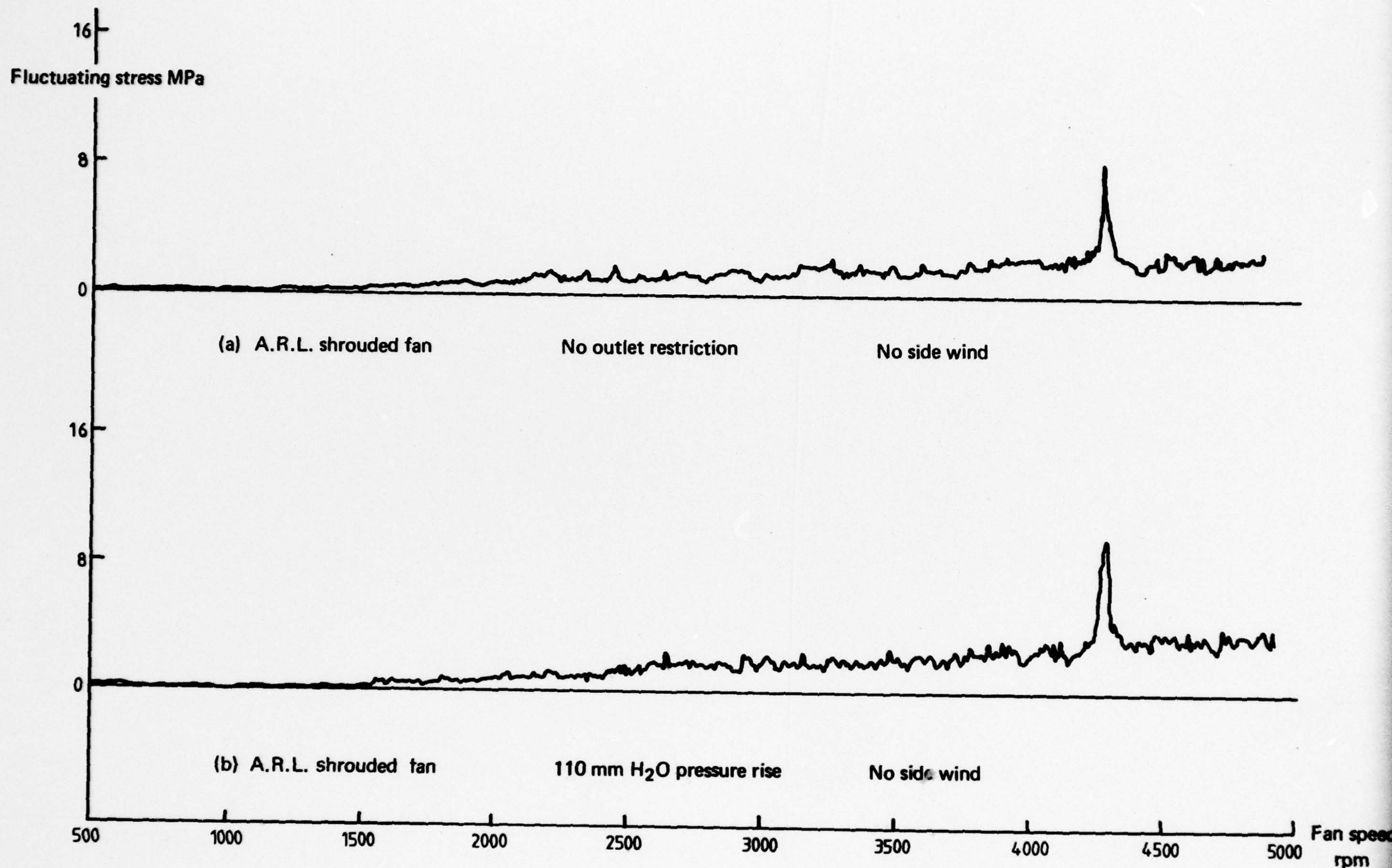

FIG. 8. FLUCTUATING STRESS IN BLADE ROOT, SHROUDED FAN

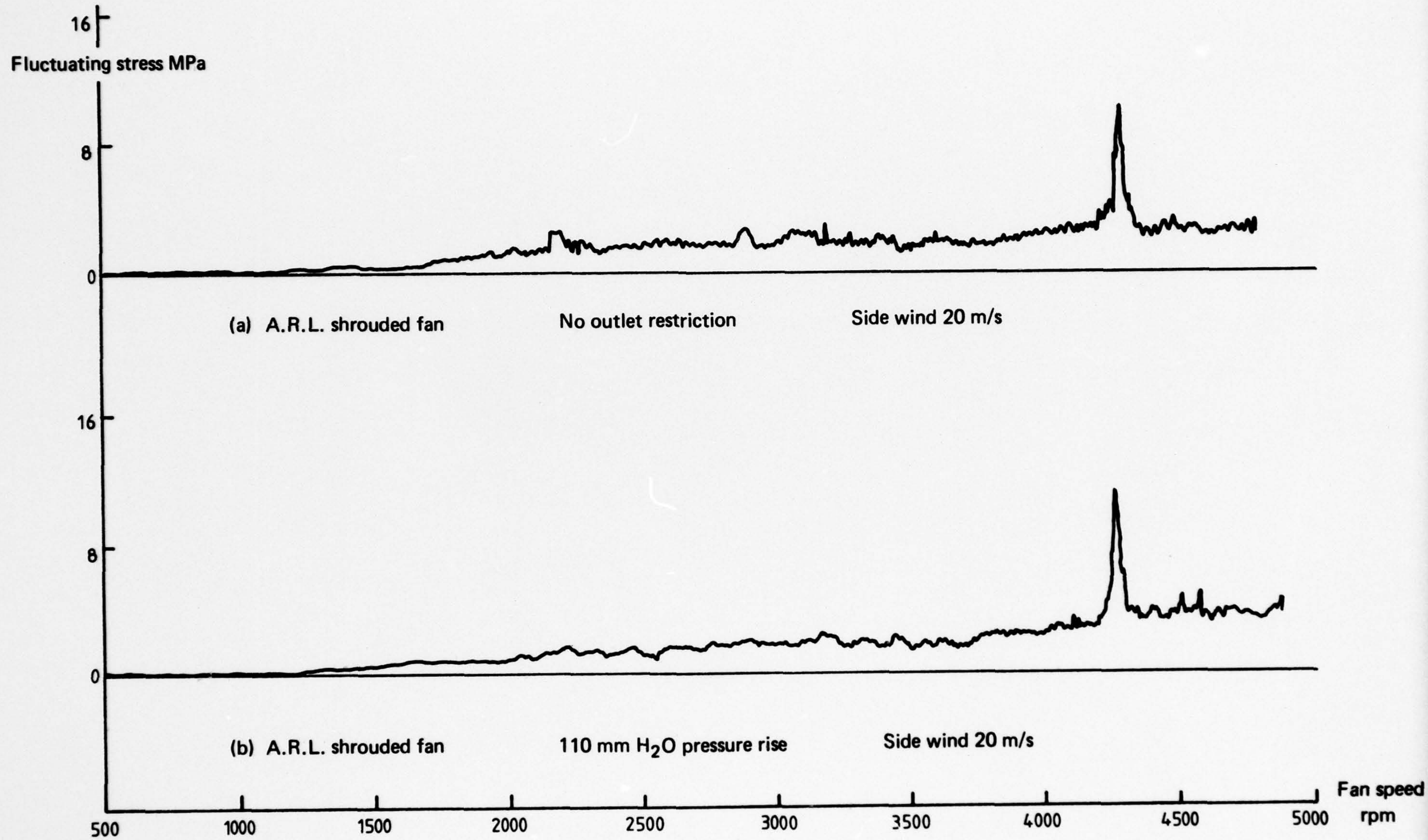

FIG. 9. FLUCTUATING STRESS IN BLADE ROOT, SHROUDED FAN

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